----- Original Message -----
From: "Bob Wands" <wands@fnal.gov
To: "Rafael Silva" <Rafael@fnal.gov
Sent: Thursday, February 10, 2005 12:21 PM
Subject: tank bottom buckling...

Rafael,

(…)

I looked at the [FEA] model [of the tank], and it calculates in-plane radially oriented stresses of as high as 160 psi compressive and 160 psi tensile. Though the stress state isn't the uniform radial compression the hand calculation assumes, we have to be conservative and assume that the plate can't take that kind of load; as we were discussing last week, we can either thicken the bottom, or (more efficiently) install some cross members on the bottom plate in direct line with the load.

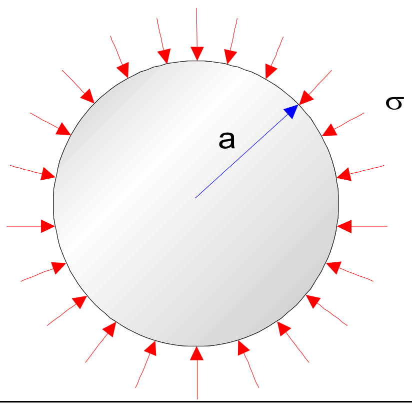
We could also look at increasing the section modulus of the thick ring [connecting the side wall with the bottom of the tank], though this might be prohibitively expensive, or just plain infeasible, if the required ring exceeds commercial capabilities, which, given the large diameter, it might.

Whichever way we go, one thing seems certain: Something has to be done.

Bob

Flare – Bottom of the tank

Isotropic circular plate under uniform radial edge compression



Status	Input	Name	Output	Unit	Comment
					Table 15.2 - Roark's Formulas
					Formulas for elastic stability of
					plates and shells.
		case	'Case_11b		Reference Number
		matnum			Material number
		matl			Material name (See material table)
	2.9E7	E		psi	Young's modulus
	0.3	nu			Poisson's ratio
					Case 11b - Isotropic circular or
					elliptical plate under uniform
					radial edge compression.
					Edges clamped
		caution1	'_		Caution Message
		caution2	'_		
	792	a		in	Radius (or Major semiaxis), a
	792	b		in	Minor semiaxis, b (b=a for circular)
	0.25	t		in	Thickness
		sigma`	3.874	psi	Critical unit compressive stress

Roark's Formulas for Stress and Strain

Total load: (2 . π . a) . t . sigma = (2 . π . 792) . 0.25 . 3.874 = 4820 lb
Unit load: t . sigma = 0.25 . 3.874 = 0.97 lb / in